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AXIAL SUBSYNCHRONOUS VIBRATION

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Mr. DeCamillo has developed performance and structural bearing analysis tools during his career, establishing design criteria used in many publications and specifications. He has patents and has authored several papers on bearing research, which is currently focused on advancing hydrodynamic bearing technology for high-speed turbomachinery applications.

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ABSTRACT

Severe axial subsynchronous vibrations (SSV) can be encountered in turbomachinery. A major concern is the potential for danger from the high levels of vibration acting on the rotor. Longer term implications are fretting and fatigue damage of the rotor components and support structures, which include the thrust bearing pivot contact areas. Although axial SSV is a concern in turbomachinery applications, little published literature can be found on the subject. A series of laboratory tests were therefore designed and performed to investigate the phenomenon in more detail.

This paper presents experimental data from a series of pivoted shoe thrust bearing tests investigating the influence of operating conditions and other parameters on axial SSV. Parameters include thrust load, shaft speed, oil flow, inlet temperature, axial clearance, pivot offset, and direct and flooded lubrication. Test results are used to evaluate techniques that have been used when attempting to resolve axial vibration issues. Additional data from thrust shoe load cells and thrust shoe proximity probes provide unique and valuable insight that allowed development of a more robust solution.

INTRODUCTION

The axial SSV under consideration is a discrete subsynchronous axial vibration of the rotor of relatively high amplitude, sometimes traveling the full set clearance of the thrust bearings. This type of axial SSV has been witnessed during mechanical tests as well as in field applications for various types of machines from small compressors to large steam and gas turbines. The occurrence is usually unexpected and can be dangerous because axial vibration is not typically monitored in industrial applications.

Three personal experiences are given here for the purpose of discussion. The first was an application for a 200 MW gas turbine that encountered an unexpected SSV during commissioning. The SSV appeared suddenly as the rotor speed passed 2700 rpm and disappeared above 3150 rpm. The frequency was 30 percent of the rotating speed with a 26 mil (0.66 mm) peak-to-peak amplitude. The axial SSV coincided with an operating thrust reversal from forward to aft at 2900 rpm.

Another experience was with a small high-speed steam turbine that encountered a 4 mil (0.10 mm) peak-to-peak indication at a frequency of 30 percent of the rotating speed as the shaft speed passed 10,500 rpm. The SSV, also coincident with a thrust load reversal, disappeared after 11,500 rpm with no trace up to 14,500 rpm.

The third experience involved a discrete sub-synchronous vibration during no-load spin tests of a high-speed compressor. Amplitudes on the order of 3 mils (.08 mm) peak-to-peak at a frequency of approximately 60 percent of rotating speed were encountered as the shaft speed passed 10,000 rpm and persisted up to the design operating speed of 12,000 rpm.

This type of axial SSV is known to be sensitive to no-thrust-load situations, which is the case in the above examples. Indications tend to disappear under load and so a common industry technique is to "preload" the thrust bearings by reducing the axial clearance. The solution, however, was not successful or repeatable in all of the above situations.

A literature search finds little published research on the subject of axial shaft SSV in turbomachinery. Gardner (1998) reports on an experimental study of thrust pad flutter investigating pivot damage of the inactive side thrust pads. The paper is often used as a reference but does not actually measure or comment on axial shaft vibration. Other techniques (gathered from personal experience and in conversations with others) include increasing the thrust bearing oil flow, modifying the balance piston, reducing the pivot offset, and creating asymmetry between loaded and slack bearings. These have also been applied with varied levels of success and failure, being based mostly on anecdotal evidence.

In order to better understand the nature of axial SSV, the author's company, Kingsbury, Inc., designed and performed a series of pivoted shoe thrust bearing tests to gather experimental data and investigate the influence of various parameters. Results are presented in this paper. The intention is to provide information of value to those involved with hydrodynamic bearings and vibration in turbomachinery.

TEST RIG AND TEST BEARINGS

Kingsbury's high-speed test rig is described fully in reference Wilkes, et al. (2000). A brief description is given here to explain how the axial vibration tests were configured.

The test rig shaft is driven by a variable speed gas turbine with a rated output of 1100 horsepower (820 kW) and a controllable test speed range of 4,000 to 17,000 rpm. The turbine is connected to the test shaft by a flexible coupling. Two identical housings external to the turbine enclosure contain the bearing components (Figure 1). Two tilting-pad journal bearings support the test shaft.

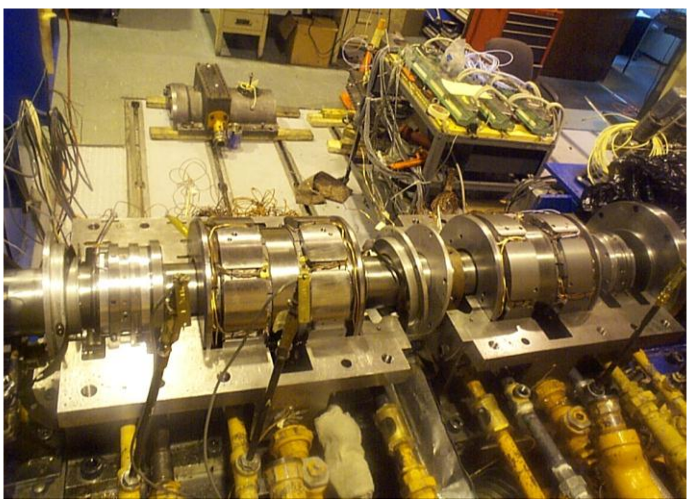

Figure 1. Test Rig Housings and Bearings.

The existing test rig was designed to test two double thrust bearings under load. Referring to Figure 2, the fixed housing adjacent to the turbine is firmly secured to the foundation while the sliding housing is restrained but is free to slide axially. Axial load is applied by means of an external hydraulic system which transmits force to the sliding housing. As a result, thrust bearings A and D are loaded against each other while thrust bearings B and C remain unloaded. A load cell mounted on the end of the sliding housing measures the applied force.

For tests investigating the influence of external thrust load, the test bearings were installed in positions A and B and a single thrust bearing was installed in position D to apply the load. For tests investigating parameters other than external thrust load, the sliding housing was bolted securely to the baseplate and the test bearings were installed in positions A and B or positions D and C, depending on the test. In all cases the test bearings were carefully shimmed to set the axial clearance and to hold the position the shaft so that no axial force is transmitted through the coupling.

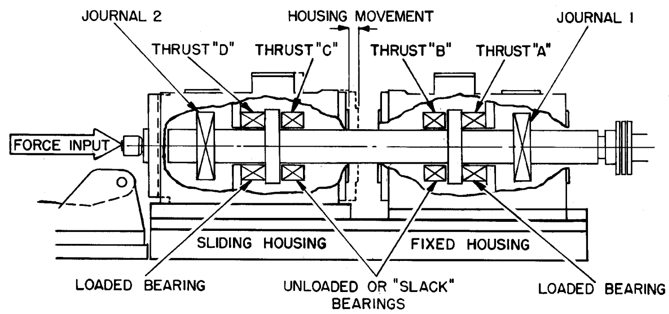

Figure 2. Test Rig Schematic.

Four thrust bearing designs were tested during the series including conventional flooded center and offset pivot 6-shoe bearings, a 6-shoe 60% offset pivot leading edge groove (LEG®) direct lube bearing, and an 8-shoe 65% offset pivot LEG® direct lube bearing which is shown in the photograph of Figure 3. All bearings have a babbitt outside diameter of 10.5 inches (267 mm) and a total bearing area of approximately 55 square inches (35484 mm^2). Parametric tests of various operating conditions and other geometry modifications were studied over the course of the series, which are explained in following sections. All tests were run with ISO VG 32 turbine oil.

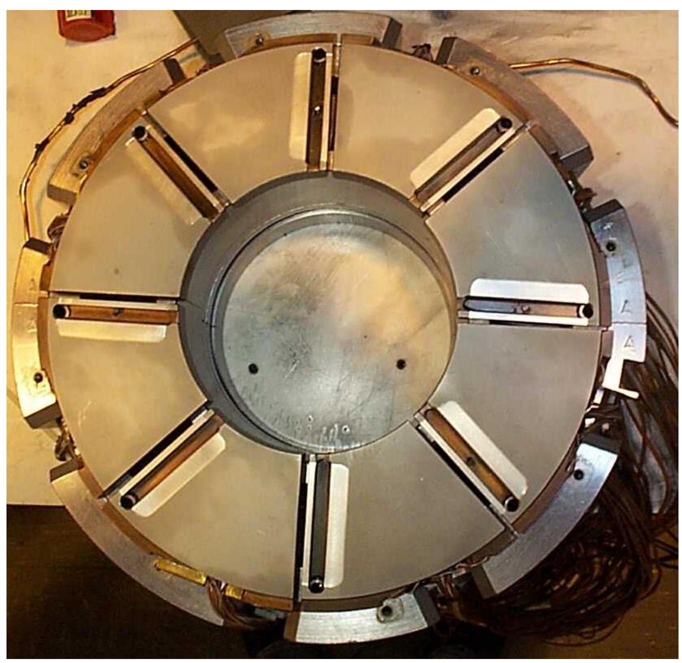

Figure 3. 8-shoe 65% Offset Pivot Direct Lube Test Bearing.

A National Instrument LabVIEW data acquisition system was used to record the test data. Steady-state measurements include shaft speed, applied thrust load, oil flow, oil supply pressure, oil supply temperature, oil drain temperature, and pad temperature. Load cells were mounted under a thrust pad on either side of the collar to monitor and record hydrodynamic film forces.

High-speed waveform data were recorded for seven vibration probes. Radial probes were mounted inboard of each of the fixed and sliding housing journal bearings, 45 degrees off top dead center, and an axial probe monitored the end of the shaft as indicated in Figure 4. Proximity probes were also mounted under a thrust pad on either side of the collar to monitor pad vibration (Figure 5). An FFT is performed to generate frequency spectrums and waterfall plots as shown in example Figures 6 and 7. The FFT and waterfall amplitude scale denotes peak rms in mils (thousandths of an inch). The backplane of the waterfall plot displays a projection of the data, and the record number is an indication of time in seconds.

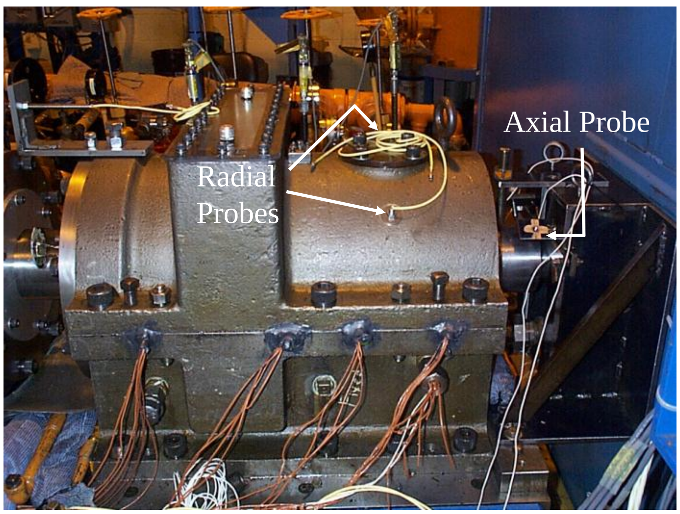

Figure 4. Fixed Housing Proximity Probes.

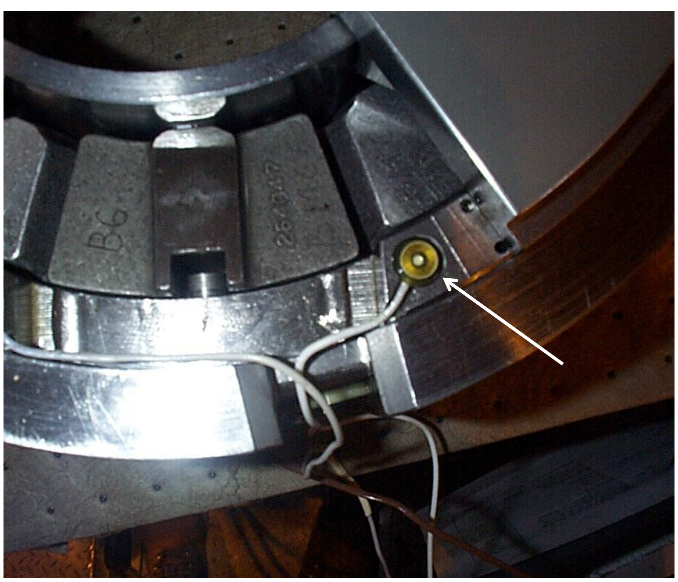

Figure 5. Thrust Pad Proximity Probe Location.

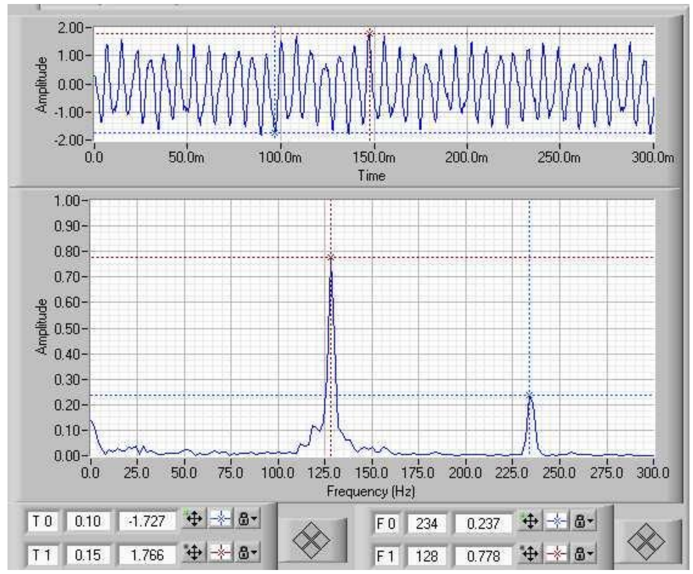

Figure 6. Example Wave Form and FFT.

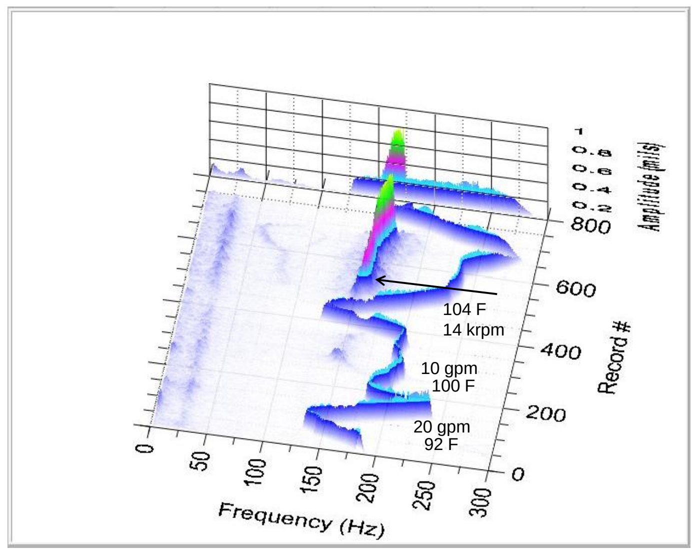

Figure 7. Example Waterfall Plot.

REPRODUCING AXIAL SSV ON THE RIG

Many operating conditions and parameters were investigated in an effort to understand the nature of axial SSV and to find a suitable method or solution to address axial SSV issues. Test rig operating conditions cover a speed range from 4,500 to 16,500 rpm and oil flow capacity up to 60 gpm (227 l/m) per bearing. Inlet temperatures from 80°F to 150°F (27°C to 66°C) and axial clearances between .004 and .019 in (0.10 and 0.48 mm) were also investigated. Representative data from the test series are provided in the following sections to relay the main points of the tests and observations.

Finding an Axial SSV Signature

The first series of tests were performed with 8-shoe 65% offset pivot direct lube bearings in fixed housing positions A and B, and a single 6-shoe 60% offset pivot direct lube bearing in sliding housing position D, backed far enough away from the collar so as not to interfere with the no-load dynamics of bearings A and B. Initial tests were run under no-load conditions while adjusting the inlet temperature, shaft rpm, oil flow, and axial clearance in attempt to find an axial SSV indication.

Many hours of test time were spent varying the parameters and the search began to seem futile until a discrete subsynchronous vibration suddenly appeared at 14,000 rpm, indicated by an arrow in the associated waterfall plot of Figure 7. Oil flow, oil inlet temperature and axial clearance at the time were 10 gpm, 104°F, and .005 in (38 l/m, 40°C, and 0.13 mm), respectively. The indication was an important aspect of the tests as axial SSV was successfully produced on a laboratory rig without many of the common sources of excitation found in turbomachinery. The success allowed continuation of a test series to investigate how operating conditions and other parameters influence the behavior of axial SSV.

Influence of Load

Beginning with the conditions that excited the axial SSV above, the influence of thrust load was next investigated by applying load using thrust bearing D. The results were fairly straight forward and consistent with field experience in that axial SSV indications disappeared when load was applied. Example results are shown in Figure 8.

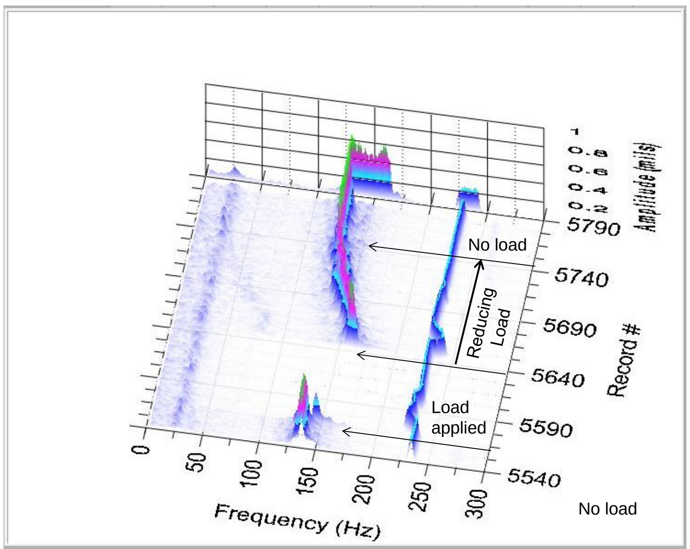

Figure 8. Application and Release of External Load.

Another important observation from Figure 8 is a shift in subsynchronous frequency between record numbers 5640 and 5740. The load was released very slowly during this interval to investigate the onset of axial SSV. A 160 Hz indication appeared as the bearing unit load passed below 20 psi (0.14 MPa) and drifted to 125 Hz as the thrust load approached zero. The results indicate that axial SSV can occur not only in no-load or load reversal situations but also in the case of low loads, which can have a significant influence on SSV frequency.

8-SHOE BEARING NO-LOAD TESTS

The thrust bearing in position D was removed, the sliding housing was bolted securely to the baseplate, and the test bearings were run under no-load conditions for the remainder of the test series.

Influence of Oil Flow

Operating conditions for the 8-shoe 65% offset pivot bearing were again set close to those that excited the axial SSV above and the oil flows to sides A and B were varied independently. In this test the axial SSV displayed a strong sensitivity to small changes in flow. Figure 9 contains data for operating conditions of 15,000 rpm, 114°F (46°C) oil inlet temperature, and .005 in (0.13 mm) axial clearance. The figure begins at record 800 with 10 gpm (38 l/m) to bearing A and 10 gpm (38 l/m) to bearing B. Arrows illustrate the results of a 3 gpm (11 l/m) increase first to bearing B and then to bearing A. The disappearance of the 165 Hz subsynchronous frequency is noticeable in the figure.

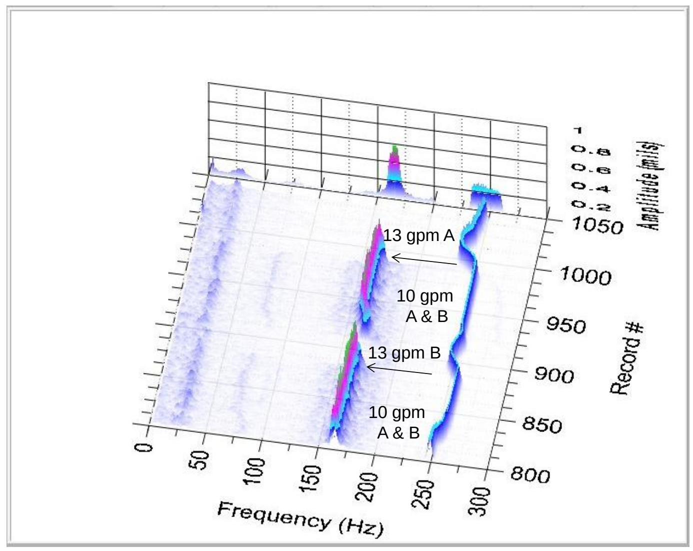

Figure 9. Influence of Oil Flow, 8-Shoe 65% Offset Bearing, 15,000 rpm, 114°F Inlet, .005 Inch Clearance.

Influence of Axial Clearance

The axial clearance for the 8-shoe 65% offset pivot bearings was incremented over the course of the test series to investigate the influence on axial SSV. The clearance was changed to a more typical value of .012 in (0.30 mm) for the next series of tests.

Operating conditions were again set to those that excited the axial SSV above, but the indication did not return. At first it appeared as though the increase in clearance had eliminated the SSV. Further investigation found that it had shifted to a different set of operating conditions. By varying speed while maintaining the 10 gpm (38 l/m) oil flow, a 45 Hz SSV was eventually encountered at 13,000 rpm. This 45 Hz SSV is indicated by arrows at the beginning and ending records of Figure 10.

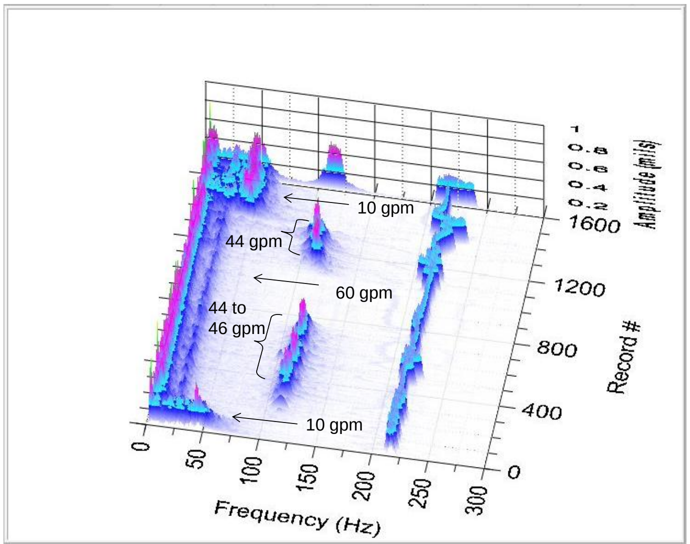

Figure 10. Flow Ramp (10 to 60 to 10 gpm), 8-Shoe 65% Offset Bearing, 13,000 rpm, 114°F Inlet, .012 Inch Clearance.

Continuing with the parametric study, oil flows to bearings A and B were varied from 10 to 60 to 10 gpm (38 to 227 to 38 l/m). As a reference, 40 to 50 gpm (150 to 190 l/m) would be a suitable flow range for field applications of this bearing size and speed. A second SSV indication at 113 Hz was encountered near 44 gpm (167 l/m), also noticeable in Figure 10. The occurrence of two distinct axial shaft SSV frequencies is similar to Gardner's (1998) inactive side pad vibration observations.

The low frequency indications (below 12 Hz in Figure 10) represent random shifts in the shaft position between the two bearings. The movement, commonly referred to as shaft shuttle, was consistent through the axial SSV and to the highest test flow. Tighter clearances (Figure 9 for example) tended to reduce the low frequency amplitudes and to increase the frequency of any discrete axial SSV indications.

Other Parameters and Comparisons

Initial investigations of axial SSV compared broad changes in parameters such as high versus low oil flow. The goal was to assess if the parameter had a negative or positive influence on axial SSV. Generalizations became difficult as comparisons seemed to produce conflicting results. High oil flow, for example, appeared to eliminate SSV in some cases, and low oil flow in others. Similar results occurred with oil inlet temperature, axial clearance, and flooded versus evacuated comparisons. The test producing the results in Figure 10 helped clarify the situation.

Figure 10 data shows a strong SSV sensitivity at 44 gpm (167 l/m) for this particular bearing configuration and operating conditions. It is apparent that the SSV indication can be eliminated by increasing or by decreasing the flow. Behavior is indicative of the excitation of a natural frequency where an increase or decrease in a parameter can cause an increase or a decrease in response depending on the proximity to the sensitive condition. The goal to assess negative or positive influence was abandoned and subsequent tests varied the parameters in more refined increments to acquire a more detailed map of the response.

Intermediate Assessment

Assessment of the test results from a resonance point of view allowed for a more consistent generalization of the test results. Parameters that tend to increase the stiffness of the bearing (i.e. lower inlet temperature, higher oil flow, tighter axial clearance) act to increase the frequency of the discrete axial SSV indications. The axial SSV is sensitive to a particular set of operating conditions which also changes with adjustment of the parameters. Resulting amplitudes therefore may be noticed to grow larger or smaller or to appear and disappear depending on proximity to the sensitivity.

6-SHOE BEARING NO-LOAD TESTS

The axial vibration series continued with tests of 6-shoe 60% offset direct lube thrust bearings installed positions D and C. Figure 11 plots axial vibration data for four speed ramps performed with 10, 20, 30, and 40 gpm oil flows (38, 76, 114, and 151 l/m) as indicated in the figure. The data is for operation with an axial clearance of .012 in (0.31 mm) and an inlet temperature of 130°F (54°C).

An axial SSV sensitivity to 30 gpm (113 l/m) is noticeable in the data for this particular bearing configuration. The indication came in as the shaft speed passed 13,000 rpm and persisted up to the maximum test speed of 16,500 rpm. Referring to Figure 11, the SSV signature can be seen to increase in frequency and amplitude from the slightest indication in the 10 gpm speed ramp to the more obvious indication in the 30 gpm ramp. No indication is apparent in the 40 gpm data.

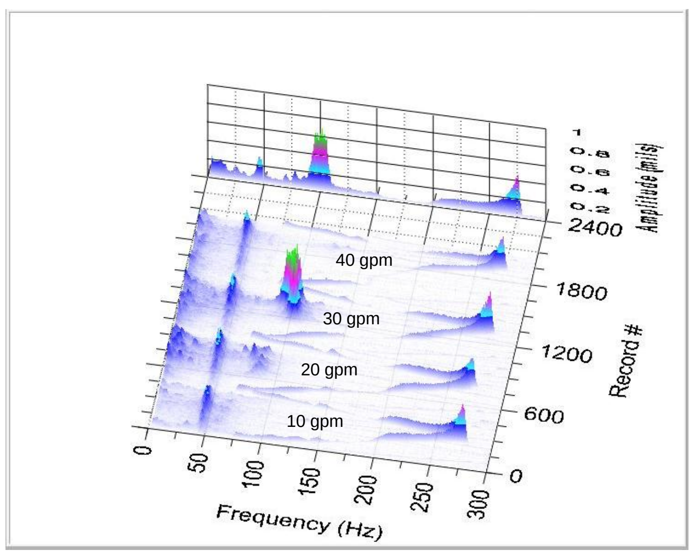

Figure 11: Speed Ramps, 6-Shoe 60% Offset Direct Lube Bearing, 130°F Inlet, .012 Inch Clearance .

As a side note, the increase in synchronous amplitude at

275 Hz is a false indication traced to the natural frequency of the axial proximity probe bracket. There is also a 45 Hz indication in the data attributed to a structural resonance. The indications are present in all 6-shoe vibration figures.

Influence of High-Speed Tapers

The test series included investigation of a leading edge taper (noticeable in Figure 3) described in detail by Wilkes, et al. (2000). In summary, high-speed turbomachinery spin tests and certain field conditions can encounter a situation of high pad temperatures at low load. The high-speed/low-load phenomenon is attributed to conditions where friction forces, which act to close the leading edge film, dominate over hydrodynamic film forces. The high-speed taper acts to keep the leading edge of the pad from closing under these conditions (Wilkes and DeCamillo, 2000).

Figure 12 contains vibration data results from tests of the 6-shoe 60% offset direct lube thrust bearings with high-speed tapers at conditions equivalent to Figure 11. Comparisons of the figures indicate an increase in axial SSV frequency and amplitude. Considering that the tests are under no-load conditions, the increase in SSV frequency suggests an increase in film stiffness, which is consistent with the function of the tapers.

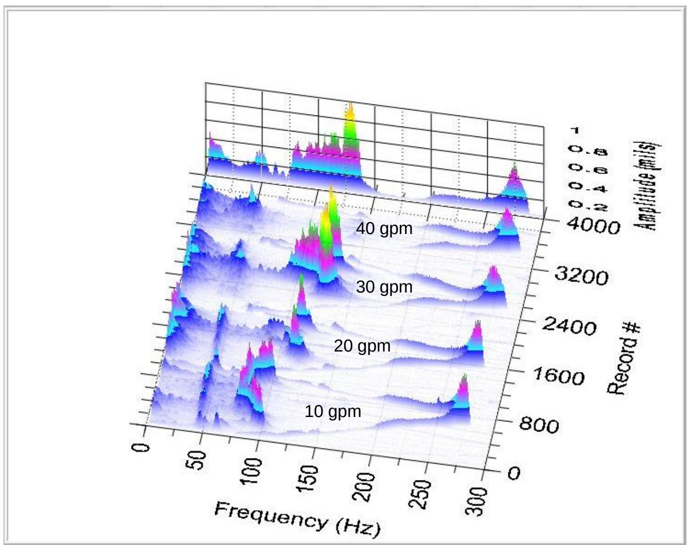

Figure 12: Speed Ramps, 6-Shoe 60% Offset Direct Lube Bearing with Tapers, 130°F Inlet, .012 Inch Clearance .

Conventional 6-Shoe Bearing Tests

The axial vibration series continued with tests of 6-shoe center and 60% offset pivot conventional flooded bearings. As above, tests include the study of the influence of axial clearance, speed, inlet temperature, and oil flow as well as the influence of high-speed tapers on the offset pads.

In the case of the center pivot bearing, it was only possible to get the faintest indication of axial SSV over the broad range of operating conditions tested. The axial SSV indications for the center pivot bearing were of very low amplitude in a frequency range between 20 and 75 Hz. Figure 13 depicts four speed ramps for the bearing performed with 10, 20, 30, and 40 gpm (38, 76, 113, and 151 l/m) oil flows.

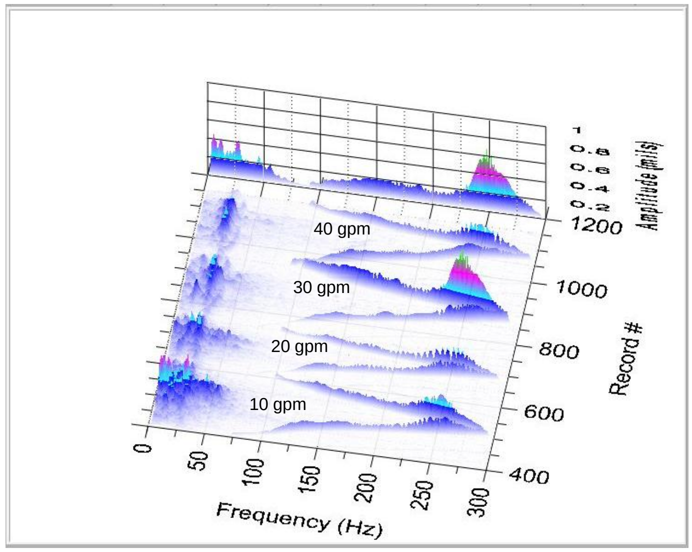

Figure 13: Speed Ramps, Center Pivot Flooded Bearing.

To better understand the low indications of axial SSV in the center pivot bearing, attention was given to the thrust shoe load cell data. The data was collected from two load cells, one mounted under a thrust shoe (Figure 14) in the bearings on either side of the collar. In the case of these no-load tests, the pad load data represents a measurement of the hydrodynamic film force acting on the pad.

Figure 14: Example Load Cell Position (Thrust Pad Removed).

Hydrodynamic Film Forces

In viewing pad load data for the 6-shoe center pivot bearing, it was noticed that the film force on either side of the collar dropped out at a relatively low speed. Figure 15 plots speed and pad load data corresponding to the axial probe vibration data of Figure 13. Figure 16 plots the same data against speed where the variation of pad load with speed and oil flow is more apparent.

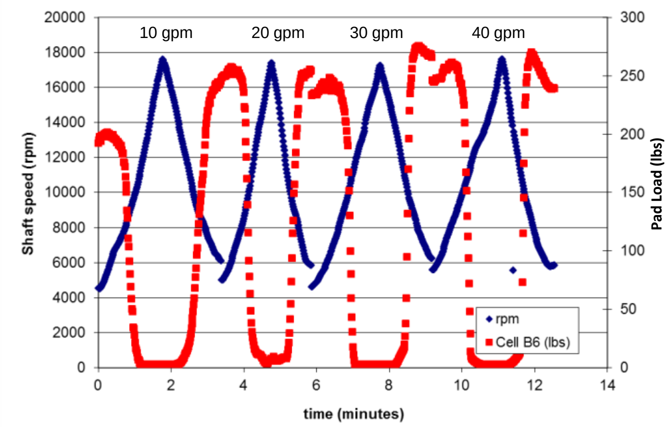

Figure 15: Center Pivot Flooded Bearing, Pad Load vs Time.

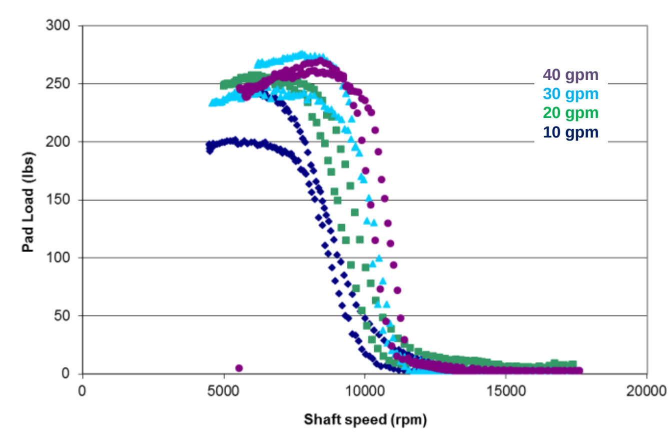

Figure 16: Center Pivot Flooded Bearing, Pad Load vs Speed.

Basic hydrodynamic theory predicts an increase in film force with speed, contrary to the test results. The data, however, actually supports the high-speed/low-load phenomenon described by Wilkes and DeCamillo, 2000. Figures 17 and 18 contain pad load data for the offset 6-shoe high-speed taper bearing corresponding to the vibration data of Figure 12. Figure 18 also shows a decreasing trend in pad loads at the higher speeds although some level of hydrodynamic film force persists which is attributed to the influence of the tapers.

While comparing and evaluating pad load data against vibration results, a trend was noticed in that conditions exciting the axial SSV to significant levels were accompanied by a certain level of hydrodynamic load. Axial SSV was not apparent in cases where the film force dropped out completely or when it was at a relatively high level. In the case of Figure 17, for example, SSV indications were high at pad loads between 75 and 150 lbs, indicated by the shaded area in the figure.

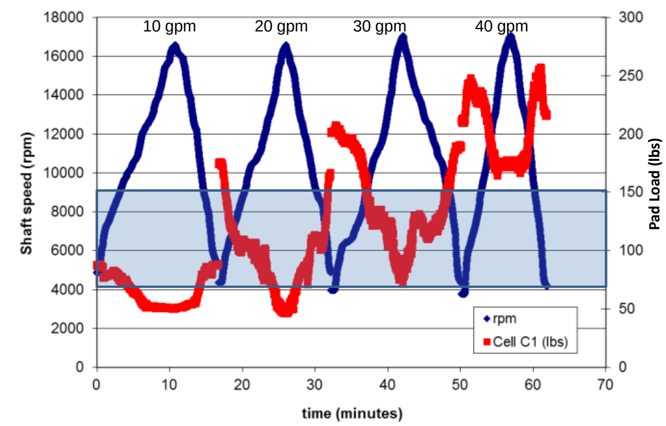

Figure 17: Offset Pivot Direct Lube Bearing, Pad Load vs Time.

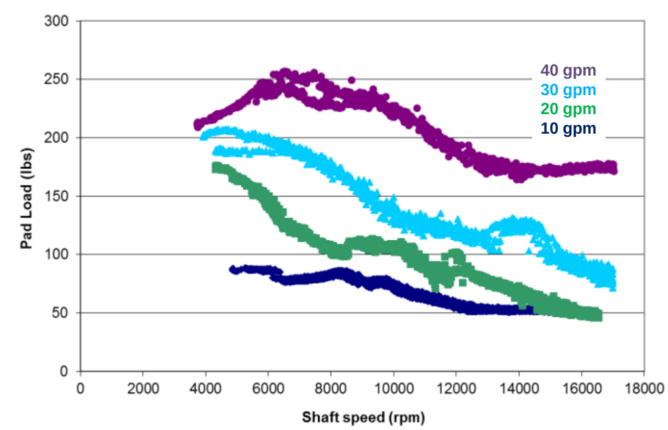

Figure 18: Offset Pivot Direct Lube Bearing, Pad Load vs Speed.

O-Ring Damper

The pad load data suggested that preloading the bearing by reducing axial clearance would eliminate SSV if the hydrodynamic force could be kept sufficiently high. This was possible in many cases, but was difficult to achieve over the broad range of operating conditions tested.

An idea was conceived to impose a relative high preload by means other than axial clearance, which is described in detail by Wilkes and DeCamillo (2006). Figures 19 and 20 show a version of the concept in the form of an O-ring placed between thrust bearing side C and its casing. The design was installed in the 6-shoe offset direct lube tapered thrust bearing and subjected to a duplicate series of tests.

Figure 19: Half Bearing Showing the O-Ring Design.

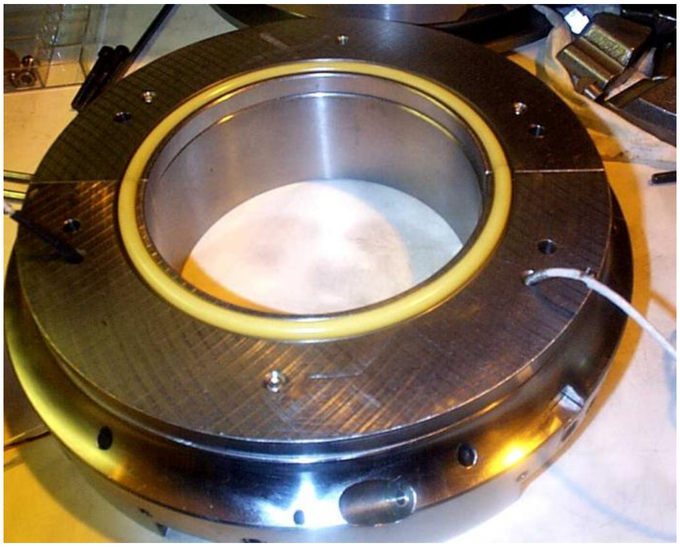

Figure 20: Assembled Bottom View Showing the O-Ring Design.

As an example, the axial vibration results of Figures 21, 22, and 23 with the O-ring duplicate the conditions of Figures 12, 17, and 18 without. Comparisons show relatively high levels of pad load and a very low amplitude axial vibration signature with no indications of axial SSV.

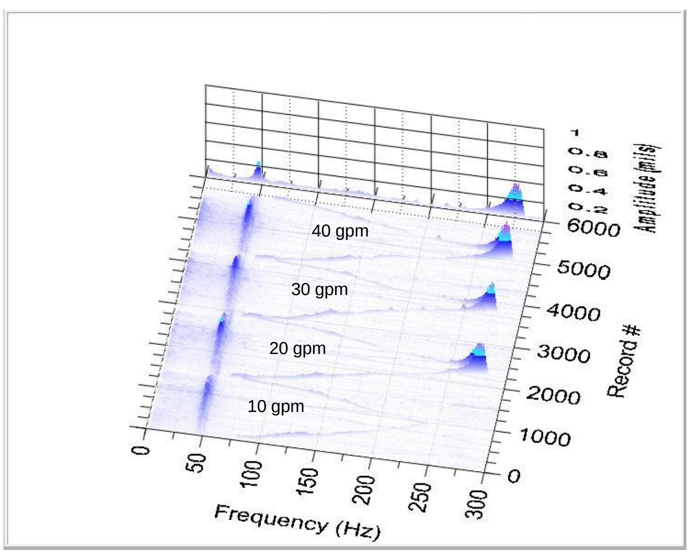

Figure 21: Speed Ramps, 6-Shoe 60% Offset Direct Lube Bearing with O-Ring, 130°F Inlet, .012 Inch Clearance.

Although the original intention of the O-ring was to raise the hydrodynamic force, the significant reduction in amplitudes over the entire range of frequencies suggests a high level of damping, derived from oil in the gap between the bearing and housing under no-load conditions. The device is therefore referred to as an O-ring damper.

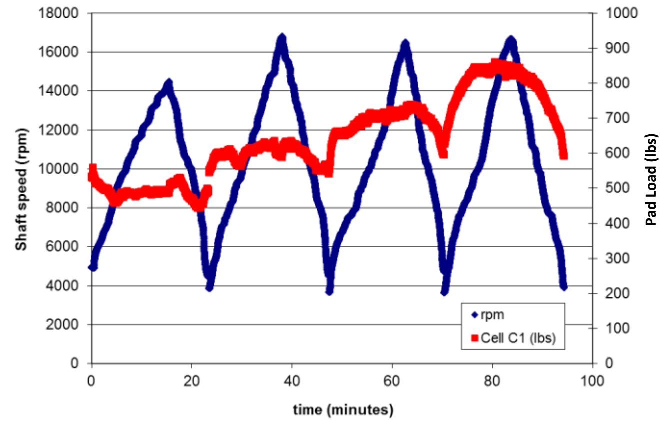

Figure 22: Offset Pivot Direct Lube with O-Ring, Pad Load vs Time.

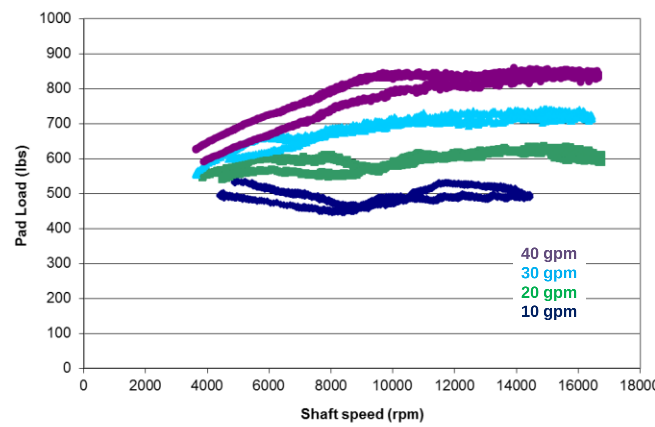

Figure 23: Offset Pivot Direct Lube with O-Ring, Pad Load vs Speed.

Pad Vibration

Pad vibration data were recorded during the course of the axial SSV test series, which is of importance as it concerns the potential for fretting and fatigue damage of the thrust bearing pivot contact areas. Figure 24 contains pad vibration data for the 6-shoe offset pivot bearing corresponding to the axial shaft vibration data of Figure 12. The frequency scale is extended to 750 Hz to capture the range of super-synchronous indications noticed in pad vibration test results.

Figure 25 is a record of duplicate conditions performed with the O-ring damper corresponding to the axial shaft vibration data of Figure 21. A significant reduction of all pad vibration amplitudes is obvious in the comparison.

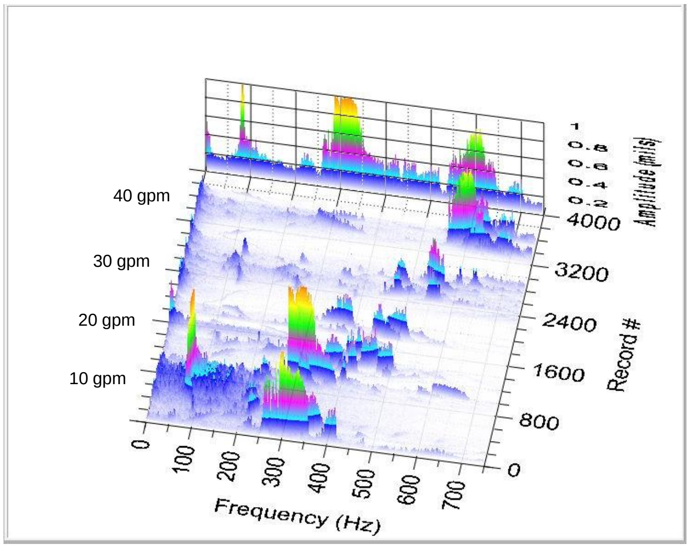

Figure 24: Pad Vibration Speed Ramps, 6-Shoe 60% Offset Direct Lube Bearing, 130°F Inlet, .012 Inch Clearance.

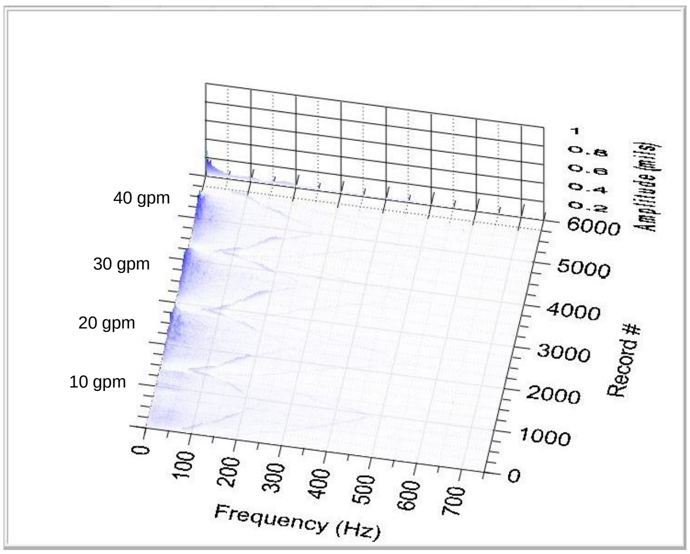

Figure 25: Pad Vibration Speed Ramps, 6-Shoe 60% Offset Direct Lube Bearing with O-Ring, 130°F Inlet, .012 Inch Clearance.

Similar comparisons of shaft and pad vibration data in duplicate tests with and without the O-ring show that the design eliminates the axial shaft and pad SSV as well as the super-synchronous pad vibrations, and generally lowers all synchronous and low frequency broadband indications. The design in Figures 19 and 20 and variations of the design have been successfully applied to resolved axial SSV issues in various turbomachinery.

DISCUSSIONS

Many of the axial SSV tests were performed to assess techniques that have been used when attempting to resolve axial vibration issues. The techniques are based on personal experience as well as solicitations of colleagues as such information was unavailable from published literature. The five techniques tabulated below have had some measure of success. The main problem is that the techniques have not proved to be reliable or repeatable:

1. Increase the oil flow
2. Tighten the axial clearance
3. Modify the balance piston
4. Reduce the pivot offset
5. Create asymmetry

The axial SSV test results suggest a resonant type behavior, which can explain the apparent lack of consistency and help provide a better understanding of the intended purpose of the technique and possible outcomes when troubleshooting axial SSV issues.

Increase the Oil Flow

Increasing the thrust bearing oil flow has proven to be a viable solution in past experience. A common mistake is to assume axial SSV is a sign of starvation, which is true in low oil flow situations. If there is sufficient flow, however, an axial SSV indication is likely a system resonance. In this case, adjustments to flow can worsen the response. The flow has to be increased (or decreased) a sufficient amount to move the resonance out of the range of operating conditions. Of all the techniques, this is the most feasible option and easiest to adjust. The greatest challenge is to assure the axial SSV does not appear over the required range of operating conditions (min/max inlet temperature, supply pressure, speed range, etc.).

Tighten the Axial Clearance

This is another viable solution proven successful in past experience. The task is more involved as the machine needs to be opened to measure and adjust the bearing clearance. Inconsistent results (higher amplitudes for example) may appear if the endplay does not push the resonant frequency sufficiently high. Another variable is that axial thrust bearing clearance is difficult to set precisely and repeatedly. If clearance is used to address axial SSV symptoms, it is important to account for possible changes in clearance over time. Opening the clearance is another possible option.

Modify the Balance Piston

This is a suitable option in many applications, again addressing the fact that hydrodynamic thrust bearings do not run well under low or no-load conditions, especially in high-speed machines. It is prudent to design for the least amount of time in no-load, low load, or load reversal operation when possible. Situations where a balance piston modification was not successful may be due to an insufficient increase in bearing load. The bearings and configuration in the axial SSV test series actually showed an increase in SSV frequency and amplitude with load before it dropped out above 20 psi.

Reduce the Pivot Offset

Test results indicate that geometry that acts to decrease the bearing stiffness tend to produce lower frequency and lower amplitude SSV. Reducing the pivot offset has been successful in eliminate axial SSV in field applications. Changing the pad offset is involved as it requires machining modifications or replacement of the pads or bearing. The solution to reduce offset also has consequences of higher bearing temperatures and lower load capacity, and may not be a suitable option in high speed machinery.

Create Asymmetry

This technique has mostly been used as a troubleshooting tool. The theory was to break the symmetry of the system or to bias the film forces by using different main and slack bearings (removing every other shoe from the slack side bearing, for example). The technique was successful in one application, and unsuccessful in another. Based on test results, it is unlikely that asymmetry is a valid theory to confront axial SSV issues. The arrangement is still a system with resonant frequencies and associated sensitivities. This technique is not recommended as a viable solution.

O-Ring Damper

The main focus of the axial SSV test series has been on shaft SSV. Discussions of the preceding techniques above are from a shaft vibration point of view and do not necessarily indicate the same behavior and results regarding pad vibration.

The O-ring damper is the result of the SSV tests series goal to understand the nature of axial shaft SSV and provide a more robust solution. It is the only technique investigated that confirmed a reliable and significant reduction in pad vibration.

The design has proved to be reliable and consistent in eliminating axial SSV. The O-ring version of the device requires periodic inspection and replacement, and the bearing design and assembly requires extra steps and considerations.

CONCLUSIONS

Axial SSV was successfully produced on a laboratory rig without many of the common sources of excitation found in turbomachinery. This allowed a series of pivoted shoe thrust bearing tests to be performed to investigate the influence of operating conditions and other parameters on axial SSV.

The test results support spin test and field experience that axial SSV is eliminated under conditions of applied load. The data more specifically indicate that the load must be of ample magnitude. Loads above 20 psi (0.14 MPa) were required to eliminate the SSV indication for the bearing size and configuration tested.

The axial SSV indications displayed a resonant type behavior. Parameters that tend to increase the bearing stiffness (i.e. lower inlet temperature, higher oil flow, tighter axial clearance, pivot offset, and high-speed tapers) act to increase the frequency. The axial SSV is sensitive to a particular set of operating conditions which also changes with adjustment of the parameters. Resulting amplitudes therefore may be noticed to grow larger or smaller or to appear and disappear.

The test results were used to critique five industry techniques that have been used when attempting to resolve axial vibration issues. The results explain the apparent lack of consistency and help provide a better understanding of the intended purpose and possible outcomes of the techniques.

Pad load cell data displayed a drop in hydrodynamic film force with speed, contrary to basic hydrodynamic theory. Based on the observations, a device was designed in the form of an O-ring damper that produced high film forces over the entire speed range, and very low amplitude shaft and pad vibration signatures with no indications of axial SSV. The device and variations of the device have been successfully applied to resolved axial SSV issues in various turbomachinery.

CLOSING REMARKS

The inference of a natural frequency raises a question as to the source of the excitation. Gardner (1998) suggests periodic pressure pulsations in the discharge annulus, which is not a possibility in the axial SSV test rig configuration. A separate study of the shaft and pad vibrations of Figures 12 and 24 displayed an intricate interaction between the supersynchronous pad vibration and subsychronous shaft vibration that may hold a key.

There is certainly a need for more research and development, experimental and theoretical, on the subject of axial SSV. In the meantime, the author hopes that the tests and data presented in this paper will provide a useful reference for topics related to axial vibration in turbomachinery.

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